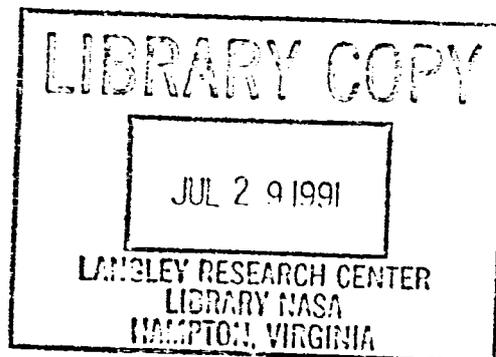


Magnetic Bearings—State of the Art

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July 1991





MAGNETIC BEARINGS - STATE OF THE ART

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ABSTRACT

E-6309

Magnetic bearings have existed for many years, at least in theory. Earnshaw's theorem, formulated in 1842, concerns stability of magnetic suspensions, and states that not all axes of a bearing can be stable without some means of active control. In Beams' widely referenced experiments, published in 1950 (Ref. 5), he rotated a tiny (1/64-in. diameter) rotor to the astonishing speed of 800 000 rps while it was suspended in a magnetic field. Earlier than that, Holmes (Ref. 41) suspended what he termed a needle in the field of a solenoid. A feedback control circuit stabilized the support; the needle was rotated to a speed of 1200 rpm.

Despite this long history, magnetic bearings have only begun to see practical application since about 1980. The development that finally made magnetic bearings practical was solid-state electronics, enabling power supplies and controls to be reduced in size to where they are now comparable in volume to the bearings themselves.

This paper attempts to document the current (1991) state-of-the-art of magnetic bearings. The referenced papers are largely drawn from two conference publications - proceedings of the first and second international symposia on magnetic bearings - published in 1988 and 1990 respectively.

PRINCIPLES OF MAGNETIC BEARINGS

Magnetic bearings come in several varieties: repulsive, eddy current, Lorentz (voice coil), and attractive. Attractive bearings have a significantly higher specific load capacity than the other types and appear to be the preferred type for use in aerospace machinery; therefore, only attractive bearings will be discussed in this paper.

Figure 1 shows schematically the horizontal axis of an attractive magnetic bearing. There is a U-shaped electromagnet on each side which attracts the bearing journal. The net force exerted on the journal is given by (Ref. 53)

$$F = \frac{1}{4} \mu_0 AN^2 \left[\frac{i_1^2}{g_1^2} - \frac{i_2^2}{g_2^2} \right]$$

where

μ_0 permeability of free space

A area of one pole face

N number of turns of wire in coil

i coil current

g gap between pole face and rotor

This equation includes some simplifying assumptions; e.g., fringing flux is neglected, as is the angle between the two pole faces. The force is obviously a nonlinear function of both coil current and magnet gap. With the arrangement shown in Fig. 1, the bearing gaps are approximated by

$$g_1 = G - x \quad \text{and} \quad g_2 = G + x$$

in which G is the gap when the rotor is centered and x is the rotor displacement from center. In order to linearize the force equation, it is common practice to operate the bearing with a steady bias current while altering the force with a control current which is of opposite sign in the two coils. Thus,

$$i_1 = I_b + i_p ; i_2 = I_b - i_p$$

and

$$F = \frac{1}{4} \mu_o AN^2 \left[\frac{(I_b + i_p)^2}{(G - x)^2} - \frac{(I_b - i_p)^2}{(G + x)^2} \right]$$

For small values of control current and small displacement, the force is well approximated by the linear relation

$$F = K_i i_p - K_x x$$

in which

$$K_i = \frac{\partial F}{\partial i_p} \quad \text{and} \quad K_x = -\frac{\partial F}{\partial x}$$

K_i is known as the actuator gain or current stiffness and is given by

$$K_i = \frac{\mu_o AN^2 I_b}{G^2}$$

K_x is the position stiffness which is given by

$$K_x = -\frac{\mu_o AN^2 I_b^2}{G^3}$$

We note that the position stiffness is negative; thus the bearing will be unstable without active control of the magnet current.

As an alternative to a continuous bias current, bearings have been designed with permanent magnets to provide a bias field (Ref. 74). These bearings have the advantage of lower current draw, but greater complexity and somewhat more difficult assembly.

Limitations. The first equation above for force may be used to estimate the maximum force capability of a particular bearing. The number of turns of wire and pole face area are limited by the physical size of the bearing. The gap G is set to allow the needed bearing deflection; in addition, it is common practice to utilize "backup" bearings, such that the load is transferred to them before the magnetic gap goes to zero. This is done in order to avoid catastrophic failure in the event of overload or electrical failure. Maximum current is limited by allowable power dissipation in the windings and also by the saturation flux density of the magnet iron. Figure 2 shows that there is a region where flux density rises rapidly with magnetizing force (which is essentially ampere-turns), but then increases at a much lower rate. The flux density at the "knee" of the curve is known as the saturation flux density. The bearing force equations presented are predicated on being below the knee of the curve; thus the saturation flux density sets a limit on the force capability. Actually, it is possible to operate with currents higher than those corresponding to saturation. As Fig. 2 shows, flux density continues to rise somewhat with increased magnetizing force; the flux density increase for some magnet materials can be substantial. The penalty to be paid for this mode of operation, in addition to less efficiency in the magnet, is greater nonlinearity of the system. Different magnet materials have different saturation flux densities (defined here as the knee of the curve), varying from 1 to 1.2 tesla for commonly used silicon iron to 1.5 to 1.6 tesla for

permeability. Since magnet force varies as the square of the flux density, a forty percent increase in flux density will nearly double the applied force.

A further limitation is set by the rate at which the control system can change the current in the windings. The magnets have an inherently high inductance which resists a change in current; the voltage necessary for a given current slew rate is

$$E = L \frac{di}{dt}$$

where L is the inductance. Thus the maximum slew rate depends on the voltage available from the amplifier. In practical terms, the required slew rate is a function of the frequency and amplitude of vibration experienced by the bearing. The combined effects of magnetic saturation and slew rate limitation are shown in Fig. 3. At low frequency, magnetic saturation limits the force attainable. At some frequency the limitations of the amplifier come into play; they become increasingly more restrictive as frequency rises. A more extensive discussion of force and slew rate limitations may be found in Ref. 53.

APPLICATIONS - ROTATING MACHINERY

A multitude of operating machines using magnetic bearings now exists, with many more under development. Some of them will be briefly described.

Field experience. A pioneer in the application of magnetic bearings was the French firm Société de Mécanique Magnétique, usually known as S2M. S2M produces attractive, actively controlled bearings. The company claims over 1000 machines using its bearings (Refs. 11,22); bearing diameters range from 25 to 1250 mm. The principal application is in turbomachinery, chiefly industrial compressors and turboexpanders; speeds range to 47 000 rpm and machine power to 26 MW. Magnetic bearings have been retrofitted to

existing machines and also provided as original equipment on new machines. The principal advantages of magnetic bearings in these applications are elimination of oil systems and lower power loss in the bearings.

Magnetic bearings have also been produced for machine tool spindles (Ref. 22). They tolerate high speed (up to 180 000 rpm) better than conventional bearings and also operate with lower vibration; the latter property results in more accurate machining. In developmental work, noncircular bores have been produced solely by appropriate control of the magnetic bearing (Ref. 58). A further advantage of magnetic bearings is that the force on the cutting tool or grinding wheel can be estimated by monitoring the current supplied to the bearing magnets (Ref. 66). Material removal rates as high as 1.6 l/min have been demonstrated (Ref. 73).

Several unusual applications have been reported. In the production of titanium powder (Ref. 48), a solid titanium billet is rotated while a plasma arc melts one end of the billet; centrifugal force carries away the resulting metal droplets. The rotating billet can have a considerable imbalance. With a magnetic bearing the billet can be allowed to rotate about its mass axis, thus virtually eliminating vibration due to imbalance. As a result, the machine can handle a larger billet and rotate at higher speed than with mechanical bearings, allowing a higher production rate of titanium powder.

In the field of particle physics, magnetic bearings are used to support a rotating "beam chopper." A 1 kg rotor operates at speeds up to 60 000 rpm while a 12 kg rotor may run to 20 000 rpm (Ref. 30).

Because of the negligible friction inherent in magnetic bearings, the rotational drag on a rotor is almost entirely due to windage (in the absence of any external load). Consequently, a

magnetically-supported rotor can be used to infer gas pressure in the surrounding medium by measuring the deceleration rate of a coasting rotor, realizing that friction is proportional to gas pressure. Such a device is used as a high-precision vacuum gage, resolving pressure to within 10^{-7} mbar (Ref. 26).

Magnetic bearings have also been used as force generators for testing of rotordynamic systems (Ref. 76). Under computer control, the time history of disturbance forces can be programmed to simulate virtually any operating condition.

Applications under development. Magnetic bearings are being considered for almost innumerable applications. Their extremely low friction, virtually infinite life, insensitivity to surrounding medium and temperature, and amenability to sophisticated computer control give them extraordinary versatility.

Because of the short life of conventional bearings in rocket engine turbopumps, magnetic bearings promise a giant leap forward for this application. Reference 29 discusses the implications of magnetic bearings for cryogenic turbopumps, while Ref. 49 reports test results for a magnetic bearing run to 20,000 rpm in a liquid nitrogen environment. In addition to longer life, magnetic bearings have shown the ability to control the dynamic response of high-energy turbopump rotors.

At the other end of the temperature spectrum, magnetic bearings are being considered for mainshaft support of gas turbine engines (Ref. 32). The principal areas needing development have to do with the high temperatures existing in engine bearing compartments. The obvious advantages are lower bearing power loss, elimination of oil systems, and possibly less stringent bearing cooling requirements.

Considerable work has been done to demonstrate the efficacy of magnetic bearings in terrestrial pumps, compressors, and turbines. The freedom from lubrication systems, with their potential of contaminating the process fluid, is an important consideration in many of these applications. Examples of machines are turbomolecular pumps (Ref. 28), canned motor pumps (Ref. 42), boiler feed pumps (Ref. 56), process and pipeline compressors (Refs. 52,67,71), nuclear reactor coolant circulators (Refs. 12,72), and air turbines (Ref. 43).

Development work has also been done on flywheel systems using magnetic bearings. The flywheels have been used for both energy storage (Ref. 81) and for spacecraft attitude control (Refs. 6,7,59). In these applications the low drag and low power requirement of magnetic bearings are essential characteristics, as well as the unlimited life of the bearing.

Magnetic bearings have also been demonstrated for use in centrifuges (Refs. 4,80) and robotics (Ref. 33).

APPLICATIONS - POSITIONING AND LINEAR MOTION

Reference 65 describes a prototype semiconductor wafer transporter. The semiconductor must be processed in vacua. Magnetic bearings allow this to occur with the magnets outside the vacuum chamber; the magnetic field readily penetrates the nonmagnetic chamber wall.

A precision linear slide supported by magnetic bearings is postulated in Ref. 23. This slide, with a range of motion of one centimeter, is expected to have a position accuracy of 1 Å.

Magnetic bearings for use in clean-room robots were tested in Refs. 33 and 34. These references also describe magnetic bearings integrated with stepping motors for use in robots.

An antenna-pointing system for a satellite is described in Ref. 38. The digitally-controlled magnetic bearing supports allow very precise (0.002°) positioning accuracy.

Magnetic levitation has long been considered in conjunction with linear motors for use in a transport system. An analysis of one such system is presented in Ref. 77.

RESEARCH AND TECHNICAL DEVELOPMENTS

Magnetic bearings can readily operate in process fluids and require no lubricant. Thus they have been considered for application as midspan dampers for flexible rotors. Experiments have shown, not surprisingly, that they are very effective in suppressing critical speed vibration (Refs. 2,14,62). In Ref. 10, the effectiveness was increased further by varying the bearing coefficients with shaft speed. The midspan damper concept has also been evaluated for use on a 900 MW turbogenerator (Refs. 16-18).

Electric current must be supplied for operation of electromagnets. Development of efficient power supplies has accordingly received the attention of researchers. Magnetic

bearings inherently have a high inductance; this requires a large voltage available to provide the required slew rate in the current. Linear amplifiers, in which the voltage not required by the bearing is carried across a transistor, were used for early magnetic bearing applications. This is a very inefficient arrangement, however, and some form of switching amplifier (e.g., pulse width modulation) is now usually considered. References 3 and 47 provide a good discussion of amplifier design.

CONTROLS

Attractive magnetic bearings are inherently unstable. Hence active controls are necessary; accordingly, control systems have received a great deal of attention.

The simplest class of controller senses the bearing position and from this commands control currents in the bearing based on the position (P part) and time derivative of the position (D part). This is known as a PD controller. So that the bearing can hold its position against slowly varying loads, an integral term may be added (I part) based on the integral over time of the position; this produces a PID controller. There are many ways to make the control system more sophisticated, such as accounting for shaft position at other than bearing locations (whether measured or inferred). The often arcane subject of control theory is invoked a great deal; since much of it is beyond the ken of this author, only some aspects of controls easily understood by mechanical engineers will be discussed here. The reader interested in going further may consult the reference list for pertinent topics. The topics to be discussed herein are collocation of bearings and sensors, analog versus digital control electronics, adaptive controls, and accommodation of rotor imbalance.

Collocation. Control theory starts out assuming that the information reported by an appropriate sensor is representative of the motion of the magnetic bearing. If, in fact, it is

not, the control currents commanded may be inappropriate. This situation can occur when the sensors are located away from the center plane of the bearing and the rotor displacement is different at the two locations. Figure 4 illustrates this. The motion sensed by the sensors for second mode vibration is out of phase with the motion at the bearing. Some of the limits when sensors and bearings are not collocated are discussed in Ref. 61.

Analog versus digital control. Early controls used only analog elements. With the wide availability and increasing speed of microelectronics, digital control systems were naturally considered for magnetic bearings. Almost unlimited versatility is available with digital controls; gains can be modified in response to any number of system parameters (rotor speed, amplitude, etc.). In contrast, once an analog controller is set up, gains can be altered only by manually adjusting potentiometers or replacing capacitors or resistors.

The bandwidth (maximum frequency capability) of digital systems is limited by the clock speed and number of operations per sensing cycle that must be performed; however, with multiple digital signal processors controlled by a master processor the bandwidth is adequate for most applications. Digitally controlled bearings are described in Refs. 8, 15, 36, 63, 64, and 78. Hybrid systems are also possible, wherein the basic controls are analog, but the gains are set in response to a digital signal.

Adaptive controls. All-digital or hybrid systems can have the bearing gains changed "on the fly," while the bearings are operating. This makes it possible for the system to adapt to operating conditions and continually minimize some objective function. Such an objective function could be, e.g., the rms average of vibrational amplitudes measured at various points along a rotating shaft. Experiments with an adaptive control system are reported in Ref. 37.

Accommodation of rotor imbalance. If a rotor is allowed to rotate about its mass center, the vibrational response will usually be fairly low. This somewhat surprising fact occurs because typical rotor support systems actually amplify the imbalance of the rotor, such that the vibrational amplitude can be many times the deviation of the mass center from the geometric center of the rotor. Allowing rotation about the mass center is the principle behind schemes to accommodate rotor imbalance. Such schemes are sometimes called "automatic balancing," but this is a misnomer since the actual balance state of the rotor is not changed. There are two methods in use. The first uses a notch filter to reduce the stiffness of the magnetic bearings at the rotational frequency. While this is effective for accommodating imbalance, the system is susceptible to other disturbances at the rotating frequency; stability can sometimes be a problem.

The second method is termed a "feed forward" technique wherein a derived signal is added that effectively cancels the bearing force called for by the response to imbalance. With this method, the bearings retain their stiffness; thus stability is not degraded. The feedforward signal can be adaptively controlled to optimize response in real time. This method is described and evaluated in Refs. 36, 46, 54, 55, and 57. Experiments have verified the efficacy of the method to substantially increase the tolerance to imbalance.

CLOSURE AND RECOMMENDATIONS

In the last decade, magnetic bearings have made the transition from laboratory curiosity to practical industrial use. Their virtues of low power consumption, controllability, and insensitivity to environmental conditions make them the bearing of choice for a continually increasing list of applications.

The principal needs for further development would seem to include

1. High temperature components (winding insulation, magnetic materials, and sensors) for use in gas turbines;
2. Methods to increase load capacity, including nonlinear digital controls for operation of "saturated" magnet materials;
3. Control systems for operation of flexible rotors through multiple critical speeds and adaptation to changing operating conditions.

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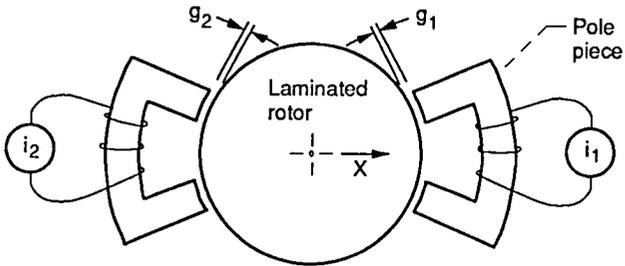


Fig. 1 Bearing schematic

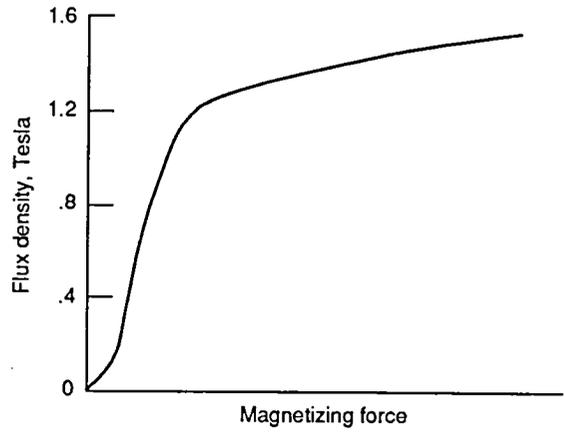


Fig. 2 Magnetization curve

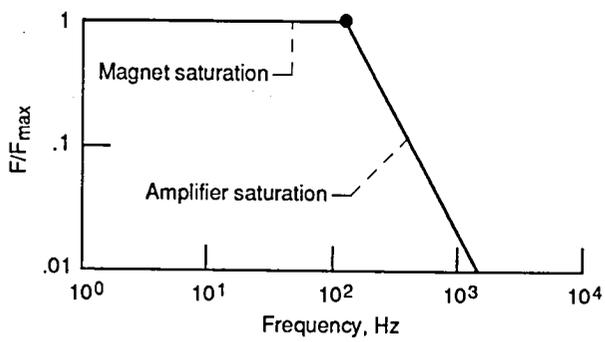


Fig. 3 Limits of bearing force

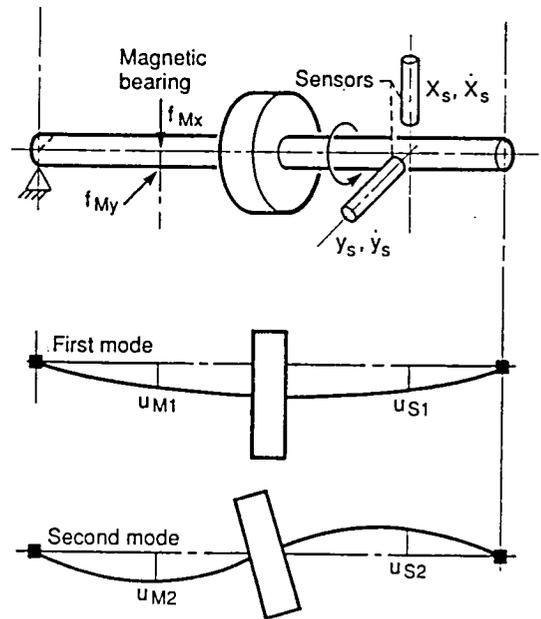


Fig. 4 Illustration of non-collocation

1. Report No. NASA TM-104465		2. Government Accession No.		3. Recipient's Catalog No.	
4. Title and Subtitle Magnetic Bearings—State of the Art				5. Report Date July 1991	
				6. Performing Organization Code	
7. Author(s) David P. Fleming				8. Performing Organization Report No. E-6309	
				10. Work Unit No. 505-63-5B	
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135-3191				11. Contract or Grant No.	
				13. Type of Report and Period Covered Technical Memorandum	
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546-0001				14. Sponsoring Agency Code	
15. Supplementary Notes Prepared for the monograph "Flight-Vehicle Materials, Structures and Dynamics Technologies—Assessment and Future Directions" to be published by the American Society of Mechanical Engineers. Responsible person, David P. Fleming, (216)433-6013.					
16. Abstract Magnetic bearings have existed for many years, at least in theory. Earnshaw's theorem, formulated in 1842, concerns stability of magnetic suspensions, and states that not all axes of a bearing can be stable without some means of active control. In Beams' widely referenced experiments, published in 1950 (Ref.5), he rotated a tiny (1/64-in. diameter) rotor to the astonishing speed of 800 000 rps while it was suspended in a magnetic field. Earlier than that, Holmes (Ref.41) suspended what he termed a needle in the field of a solenoid. A feedback control circuit stabilized the support; the needle was rotated to a speed of 1200 rpm. Despite this long history, magnetic bearings have only begun to see practical application since about 1980. The development that finally made magnetic bearings practical was solid-state electronics, enabling power supplies and controls to be reduced in size to where they are now comparable in volume to the bearings themselves. This paper attempts to document the current (1991) state-of-the-art of magnetic bearings. The referenced papers are largely drawn from two conference publications—proceedings of the first and second international symposia on magnetic bearings—published in 1988 and 1990 respectively.					
17. Key Words (Suggested by Author(s)) Magnetic bearings Surveys			18. Distribution Statement Unclassified - Unlimited Subject Category 37		
19. Security Classif. (of the report) Unclassified		20. Security Classif. (of this page) Unclassified		21. No. of pages 26	22. Price* A03

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